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Gascooler optimal pressure revisited

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ABSTRACT

The quest for finding the optimal relation between the power consumed by the compressor and the performance of the heat exchanger rejecting the heat, either to the ambient or utilised for heating purposes has been the subject for many studies not least when CO_2 is used as the working fluid.

The practical embodiment in commercially available control systems is usually incorporating a curve to minimise the compressor load for a given set of operating conditions approximating the optimal relation between the pressure and the temperature at the outlet of the heat exchanger regardless of if it is cooling or condensing CO_2 gas.

However, this curve comes short of optimising heat pump or heat recovery operation when a given temperature is needed for supply to, and under varying temperatures returned from, a heating system. This is analysed as an first approach by a simple model for off-design operation conditions for a given heat exchanger. As a second approach a manufacturer calculation software for plate heat exchangers is used.

For a large range of operating conditions, we found that the optimal pressure seems to be considerably higher than the pressure resulting from the commercial controls systems. This is also the case for an air-cooled heat exchanger rejecting the heat to the ambient modelled in a freeware from NIST and this was a surprise as it was believed that the controls were developed for these applications.

Results are presented that indicate the potential improvement in both efficiency and capacity ranging from app. 5% to 20% for subcritical to trans critical operating conditions. For the air-cooled gascooler the highest potential for improvement seems to be for operation in vicinity to the critical point.

Keywords: Refrigeration, CO_2, gas cooler, heat rejection, COP, optimal pressure, Energy Efficiency.

1. INTRODUCTION

How to specify, select and operate the gas cooler/ condenser (in the following called the gascooler for short) for the heat rejection system of a CO_2 refrigeration system or heat pump are becoming more and more important as the number of systems is steadily increasing all over the world.

One aspect of this is how to use the selection program for compressors and heat exchangers to find the optimal match in both first cost and operating economy and how these components and their operating set points are commissioned in the chosen control system.

From previous work with gascooler operation and design for an air-cooled condensing unit ((Heerup & Borup, 2018) and a supermarket hydronic heat reclaim case (Heerup, 2019) a wish emerged for a better understanding of what is understood by optimum pressure. It became clear that the optimum pressure understood as the default setting in the controls system was not in all cases sufficient to meet the required target temperature, capacity, nor efficiency.

In Figure 1 is an illustration from a control system manual showing the reference curve defining the optimum control pressure. Quote from the manual (Danfoss, 2021):

“The controller is pre-programmed to follow the optimal COP from the pressure/enthalpy chart. The top point is defined at 100 bar, 39°C. (Optimal theoretical COP is achieved at the curve that passes through 100 bar and 39°C. The point of intersection can be changed by setting a value other than the default).”

Note that the reference curve covers a seamless transition between the subcritical and the trans critical state. More information can be found here (Prins, 2016).

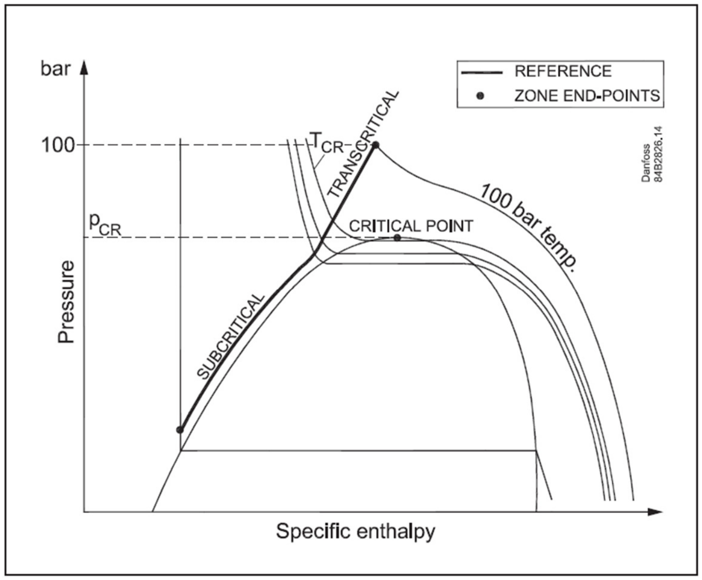


Figure 1: The reference curve for controlling the high pressure.

For CO_2 installations in the northern hemisphere without heat recovery subcritical operation constitute most of the operation time. There are several studies that investigate for various refrigerants both in theory and in practical experiments how the system efficiency relates to subcooling at the outlet of the condenser e.g. (Pottker & Hrnjak, 2015). Most are related to charge optimisation, but ((Koeln & Alleyne, 2014) (Hervas-Blasco et al., 2018) describes important increases in efficiency by enabling the control of the subcooling by introducing an extra valve to this end.

The important fact here is that on the majority of commercial trans critical systems this valve is already in place in the default configuration as a high-pressure valve and is used for optimisation. The high-pressure valve is needed in the trans critical state to actively control the pressure as the temperature and the pressure are independent parameters in contrast to the subcritical state where the relation is defined by the saturation curve, but even in the subcritical state the high-pressure valve is needed to control the subcooling as the receiver works as an open flash tank where the pressure is controlled independently by a gas bypass valve (see further system description in section 2.2).

The finite heat exchanger which has a cost related to footprint, sound level and secondary flow profiles and still needs to be operated in the most economical way is what motivates this paper. Footprint and sound level are interrelated properties for air cooled gascoolers. When the footprint is limited the face area is also limited and for a given capacity the air flow will need to be higher. As a consequence the air side pressure drop increases as well as the space available for fans decreases. This implies higher fan speeds and thus higher sound level. And vice versa: If the sound level needs to be limited, the air speed (face velocity) and the air side pressure drop will also need to be limited and this necessitate a larger face area.

This impacts the temperature rise of the secondary flow which is also the case for heat recovery on hydronic systems where certain temperature requirement must be met. All this will impact the thermodynamic properties for the heat exchanger and especially the mean temperature difference (dtm) between the two fluids. This will be highlighted based on two cases, one for an air cooled and one for a brine cooled gascooler. The study is preparatory to investigation in gascoolers and their associated controls by advanced models employed in the project Digital Twins aiming at better operational efficiency and will be concluded with field tests.

2. GASCOOLER AND SYSTEM PROPERTIES

2.1. Simple gascooler model

In one of the cases investigated the gascooler is designed as a plate heat exchanger in a water-to-glycol heat pump recovering heat from air conditioning to hot tap water in a hotel. The original selection of the heat exchanger is based on 90 bar inlet pressure and 35 °C leaving temperature. This pair of values is often used as reference for the tabled performance for compressors and is relatively close to the optimum pressure (86,4 Bar) stated by the compressor manufacturer, owing to the low pressure drop below 0,4 bar at full load. To visualise the temperature profiles for the primary and secondary flows a simple model was made in EES (Engineering Equation Solver) enabling the representation of the gascooler in a h-T diagram in Figure 2.

$$C = U \times A \times dtm \qquad \text{Eq. (1)}$$

Where C [kW] denotes the capacity, U [kW/m²K] the thermal transmittance, A [m²] the heat transfer surface and dtm [K] the mean temperature difference. The UA value is a characteristic parameter for a heat exchanger and is here assumed to be constant and so are the dtm at 13,8 K. Note that the mean logarithmic difference cannot be used here instead of dtm and would lead to wrong conclusions as it would require that the specific heat capacity for CO_2 was assumed constant and the temperature profile a straight line on Figure 2 as is the case for the hydronic side.

The analyses performed with the model shows that minimising the approach by raising the pressure results in the highest COP_h (efficiency for heating) and this also results in moving the pinch to the outlet of the CO_2 coinciding with the approach.

The operating point can be seen to the left part of Figure 2 and does not include the pressure drop in the gascooler. To meet the characteristics of the building the wish emerged to meet higher inlet and leaving water temperatures which is shown to the right in Figure 2. The CO_2 inlet temperature and the compressor absorbed power was calculated with the isentropic and volumetric efficiencies modelled by curve fitting based on compressor manufacturer data (Copeland).

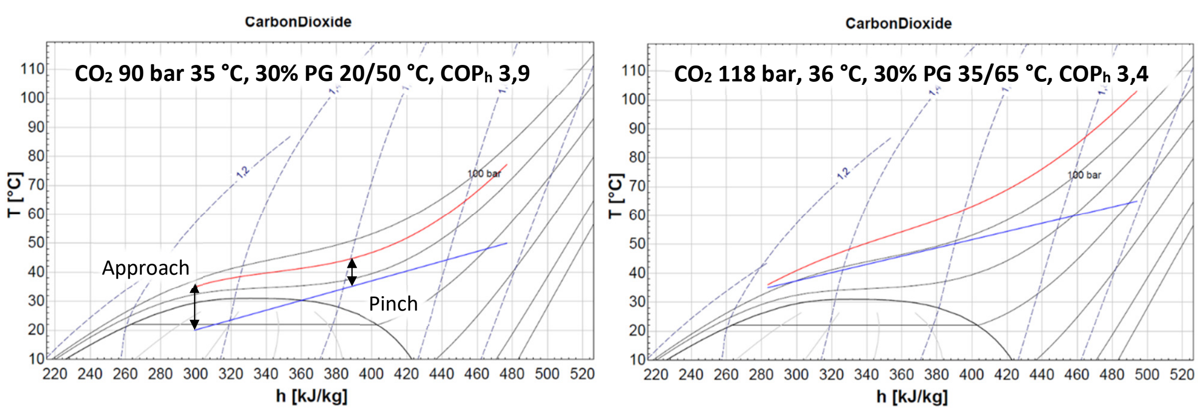


Figure 2: Gascooler temperatures in h-T diagram

However, due to the simplicity of the model the COP_h keep increasing with the pressure until the approach reach zero and this is clearly not realistic for physical systems. Further, even if the value for the COP_h would satisfy under the circumstances it is not possible to run the system at the 118 bar indicated. The reason is that it is too close to the design pressure for the heat pump, and this is also why it was decided to fit an extra heat exchanger in series with the original to reduce the required pressure and increase the efficiency as is analysed in section 2.4.

2.2. Compressor manufacturer selection tool

Compressor manufacturers provide very useful tools for calculating and visualising of operating performances. An example is show in Figure 3, where a part of the online interface showing the system components and the h-log(P) diagram is copied (Bitzer). The brown numbered lines are the original and the lines for differing operation conditions are added by the author. The input is 90 Bar and 35 °C leaving temperature as described above. It is not possible in the tool to ascribe a pressure drop to the system piping and more important neither to the gascooler. In a real system with a pressure drop the gascooler would be represented by the (red) sloping line (exaggerated). The pressure drop will effect a lower enthalpy difference and thereby lower system efficiencies. For this reason the compressor online selection tool is used to determine the mass flow and absorbed power based on input for the inlet and the outlet condition of the compressor and the resulting capacities and efficiencies are based on the output from the gascooler models described in section 2.3 and 2.4. For the calculations in section 2.4 the air-cooled gascooler in Figure 3 is substituted by a plate heat exchanger and instead of a single compressor there will be three in parallel. The evaporation temperature is +5 °C instead of -8 °C and the saturation temperature in the intermediate receiver is raised from 5,3 °C to 10 °C.

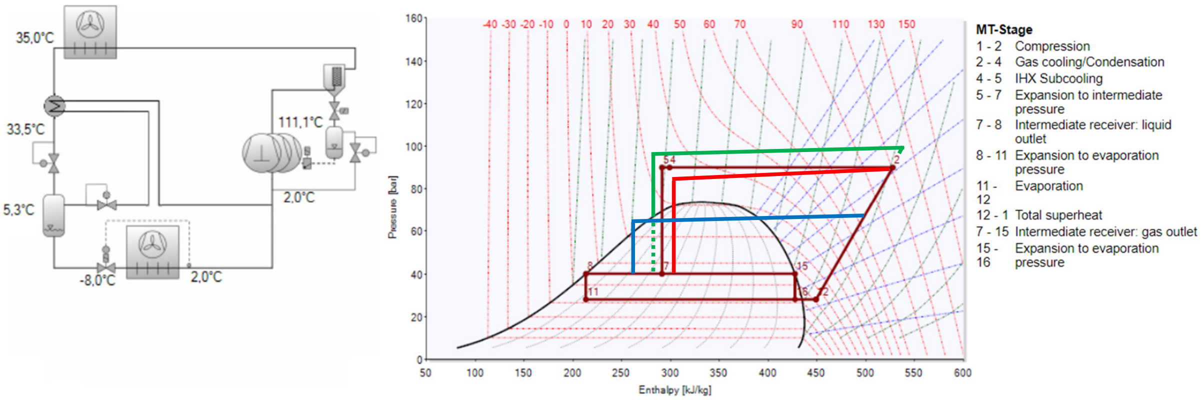


Figure 3: System layout and representation in h-log(P) diagram

The calculations show that, if the inlet pressure is increased as shown by the upper (green) line the following will happen:

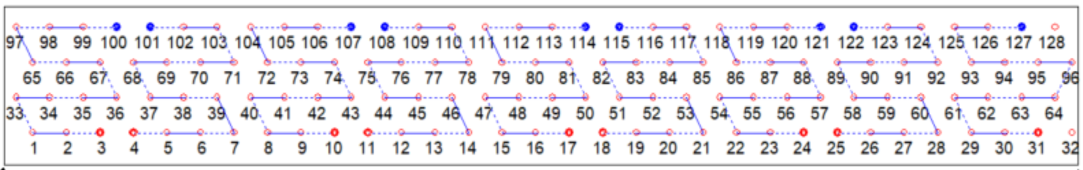
a) The mass flow will decrease with the volumetric efficiency due to the higher pressure-ratio.
b) The compressor absorbed power will increase.
c) The gascooler inlet temperature will increase.
d) The approach will decrease as described in 2.1.
e) The mean density will increase.
f) The pressure drop in the gascooler will decrease due to the lower mass and volume flow.
g) The enthalpy difference will increase as both the change in temperature and pressure at the outlet and the inlet to the gascooler will contribute to this.
h) Both the capacity and the efficiency will increase as the improvement in gascooler performance will be higher than the increase in absorbed power.

If the pressure is increased even further the efficiency will pass its maximum but the capacity will continue to increase. There will naturally be a limit to this, but it seems that it will be the system maximum operating pressure that will be the hard constraint.

For subcritical operation (the lower blue line in Figure 3) there will also be an optimum when the pressure and thus the subcooling are increased, but an important practical aspect will be the influence of the tolerances of the temperature and pressure transmitters used. If these offset the measurement of the subcooling governing the control of the high-pressure valve (positioned at the upper left of the system diagram) there is a risk that the outlet of the gascooler will contain non-condensed gasses and higher enthalpy. In the diagram the conditions for the outlet will move to the right side of the saturation line coinciding with the punctuated green line. In this case the resulting loss of efficiency by bypassing gas directly from the gascooler to the compressor could be mitigated by running with a slightly higher pressure setting as the potential losses associated with this is believed to be lower. This is supported by field experience where lower energy consumption for some cases has been reported despite a higher pressure setting.

2.3. Air cooled gascooler model

The software EVAP-COND is used to model a gascooler to represent a specific model specified by a Danish supermarket chain. This software enables the specification of inlet temperature and (mass) flow of air and mass flow, pressure, temperature of CO_2 for a given coil. The heat rejection capacity is within 4% compared to the original selection data calculated by the gascooler manufacturer for both sub and trans critical operation. However, the nominal load designated by 1,0 in the following analysis is 17% higher due to the chosen compressor in the selection software. The configuration of the circuits is shown in Figure 4.



The arrow show the air flow direction. Blue dots are inlets and red dots are outlets for CO_2.

Figure 4: Configuration of the 9 circuits in the modelled gascooler

The results of the calculations are shown in Figure 5. The designation of the curves is explained by the example of the full bold line to the left of the figure: “33 1,0 1,00” where “33” is the air inlet temperature, “1,0” is the evaporator load and “1,00” is the fan speed. The corresponding temperature approach between air on and CO_2 out of the coil is shown by the corresponding but thinner line type. The evaporator load is held constant by reducing the speed of the compressor inverter drive except for the data “33 1,15 1,00” where the compressor operates at full speed at 70Hz reaching 15% higher evaporator capacity than reference at peak efficiency (18% higher at peak pressure). The optimum COP for part load “33-0,5-1,00” is lower than for full load “33 1,0 1,00” this is due to the reduced efficiency of the compressor at lower speeds. The curves for trans critical operation originate at the pressure specified as “optimum pressure” by the compressor manufacturer for the CO_2 leaving temperature. The curves for subcritical operation originates at the pressure corresponding to between one and two degrees of subcooling, as the compressor speed can only be adjusted by increments of 1 Hz.

In all cases the approach is reduced as the pressure is increased and that for the efficiency which is designated by COP_c+f as it includes the gascooler absorbed fan power there is a maximum. The fan absorbed power at full speed is 2,2% of the compressor absorbed power at the design load.

$$\text{COP\_c+f} = \text{evaporator capacity} / (\text{compressor absorbed power} + \text{fan absorbed power}) \qquad \text{Eq. (2)}$$

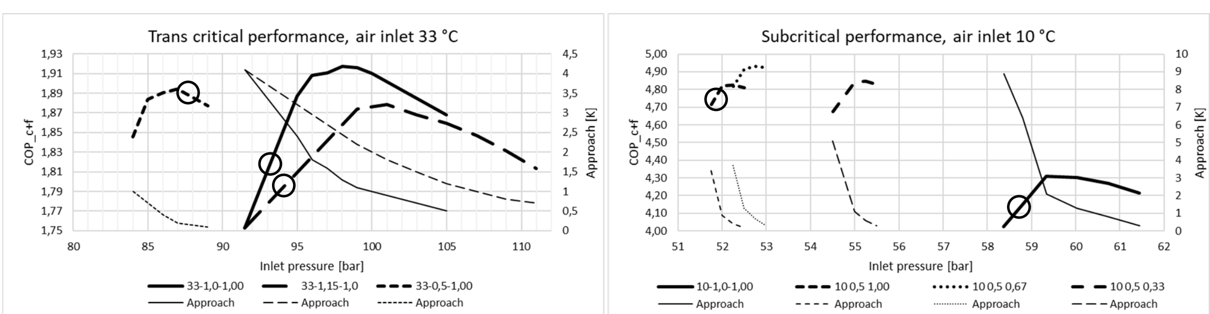


Figure 5: Trans critical and subcritical performance of air cooled gascooler at full and part load.

The optimal pressure changes with both the load and the air flow and this can make it difficult to evaluate at commissioning if the default setting of the control system is realising the best efficiency. The default setting of a commonly used control system is shown with circles. For trans critical operation the optimal efficiency is only met at low load on the gascooler whereas for the higher loads the gascooler will not perform well. If this is observed in the field it might be concluded that the gascooler is too small if it is recognised at all. For subcritical operation (only two circles shown) the setting shown corresponds to 3K subcooling, but the difference is not that big. The danger here is, that due to the limitations in accuracy, as previously pointed out, to have no subcooling and gas passing and loading the compressor.

To visualise the thermal relationship internally in the gascooler see Figure 6 which is based on the calculations in EVAP-COND and visualised with the help of EES. The linear black sections between the small circles represent a pipe and the inclined line below the increase from mean inlet to mean outlet temperature of the air cooling the pipe, so that the air first passes one row of pipes and then the next row and so forth. The air can with less accuracy be represented with a single line as shown in section 2.1. The air temperatures are show simplified as mean temperatures for air on and off the pipes whereas in reality both temperatures would change along the length of each of the gascooler pipes.

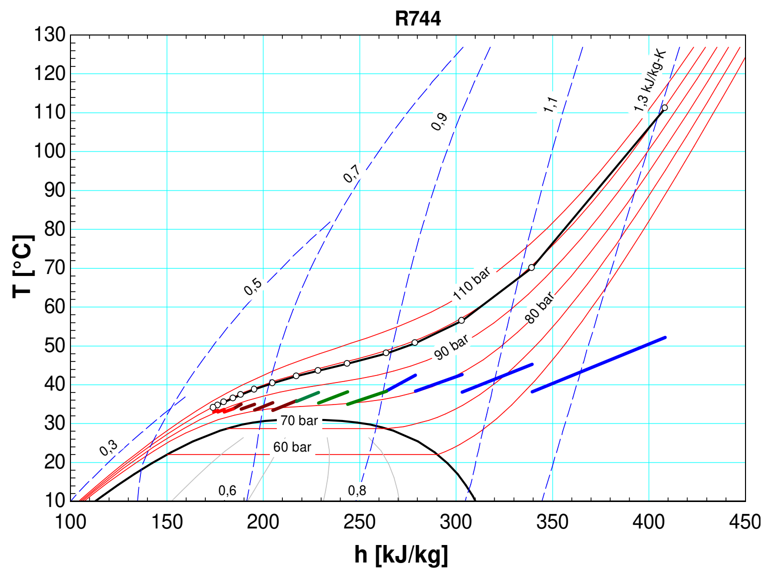


Figure 6: Diagram showing one gascooler circuit in h-log(T) diagram at 99 bar inlet pressure and 1,1 K approach

In Figure 7 is shown in a h-log(P) diagram the gascooler working lines for the curve “33 1,0 1,00” from Figure 5. The full line is representing the same working conditions as in Figure 6 at 99 bar inlet pressure and 1,1 K approach. It is clearly seen how the gascooler exit temperature approaches the air inlet temperature at 33 °C with the higher inlet pressures and thus increases the enthalpy difference.

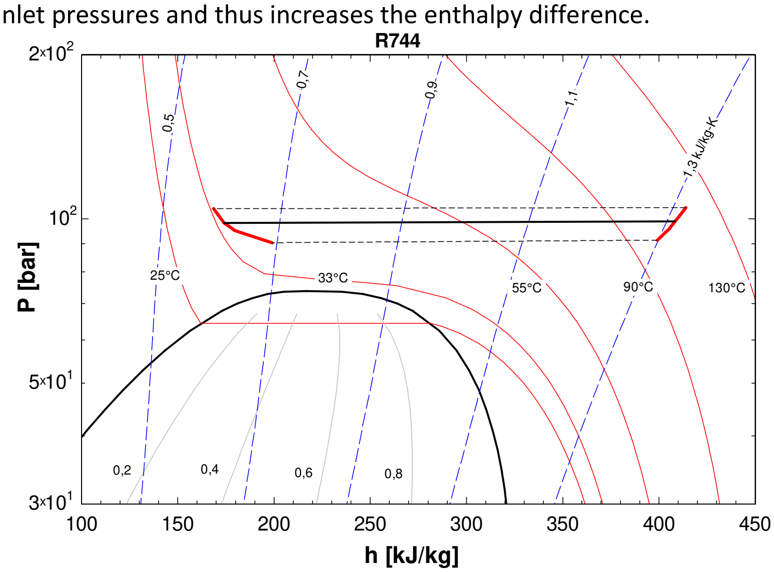
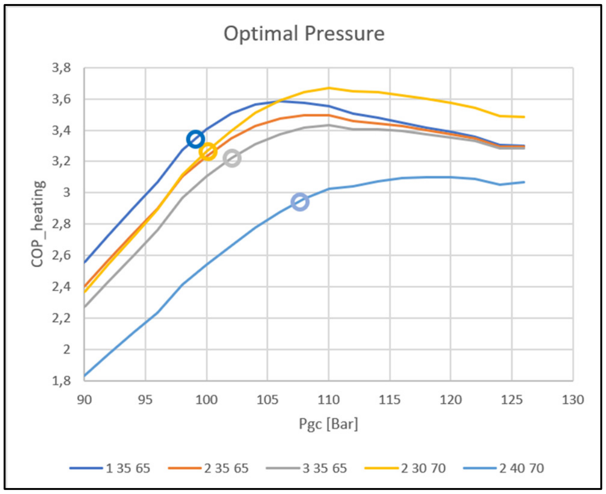


Figure 7: Diagram representing the “33 1,0 1,00” curve in the h-log(P) diagram

2.4. Glycol gascooler calculation

For a gascooler cooled with a glycol solution for heat recovery similar calculations have been made using a plate heat exchanger calculation software from the manufacturer. In this case it only make sense to analyse the trans critical operation as the temperatures on the secondary side are above the trans critical temperature. The results can be seen in Figure 8 where e.g., “1 35 65” designates one compressor engaged, entering glycol temperature 35 °C and leaving water temperature 65 °C. Also, here the approach becomes smaller as the pressure increases and an optimum for the efficiency COP_heating can be seen. Note that dTgc here is the same as the Approach Temperature. The default setting of the controller is marked with circles. Again, the pressures appear to be too low to obtain the optimal efficiency.



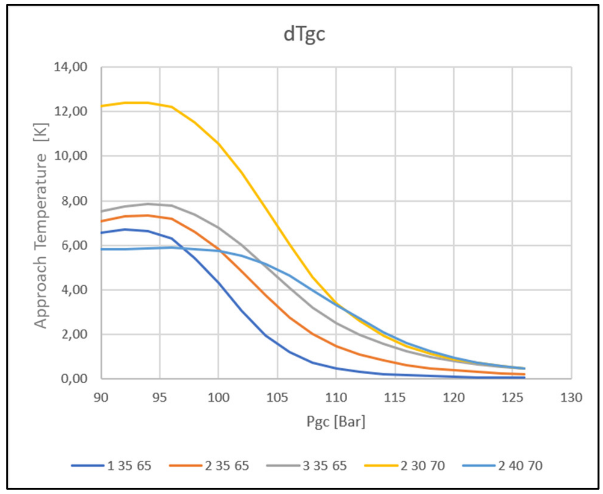


Figure 8: Optimal pressure and associated temperatures for a gascooler executed as a plate heat exchanger for heat recovery

2.5. Discussion

The results of the present study should maybe not have been a surprise as already more than 10 years ago (Cecchinato et al., n.d.) concluded:

“Controllers presently employed on the market are based on a PID feedback loop that aims at tracking a reference set-point generated by an approximated equation. This can eventually configure an adaptive control architecture but, being the solution not the optimal one, the system overall energy efficiency is penalised. This control technique might be efficiently used only if the implemented approximated solution is obtained by considering the operating characteristics of the specific refrigeration unit. In order to obtain such a tailored equation, an experimental or simulation campaign is needed for each unit, thus requiring dedicated tuning for each controller.”

However, it seems that these findings are only very slowly being recognised by the engineers in the field of refrigeration.

Above conclusion is based on a very detailed examination of different methods targeting the optimum pressure and the use of numerical models for both heat exchangers and two refrigeration systems. The first for a heat pump and the second for a supermarket system. The heat pump case exhibits similar variations as examined here.

The present study support above conclusion, however, the gascooler design modelled for the simulation of the supermarket system has a large surface relative to the load and a very low pressure drop. In fact, the surface to capacity ratio is 25% larger, the airflow 3 times larger and the pressure drop only 18% of the gascooler used in the case analysed in this paper which is based on a specification from a Danish supermarket chain. It is believed that this oversizing obscure the influence of the heat exchanger geometry on the optimum pressure. It is also believed that the reason for this is that the larger the heat exchanger and the higher the secondary flow the more the gascooler resembles the ideal infinite heat exchanger where the CO2 outlet temperature is independent of the inlet pressure.

In a later study (Noeding et al., 2019) a zero gradient control method for controlling the high pressure is presented which could be promising as it can adapt to changing conditions such as fouling and include the power consumed by the fan for the air-cooled condenser. However, no data has yet been published covering the application on a commercial control system for e.g., a supermarket refrigeration system.

3. CONCLUSIONS

Based on the present analysis it is recommended that selection procedures for gascoolers should include the effects of the pressure drop, inlet pressure for both design load as well as part load or of-design conditions which for heat pumps also include the required temperature profile for the secondary fluid.

It seems possible that it could be beneficial to use higher operating pressures for the relatively low number of hours at high ambient temperatures and to benefit from higher efficiencies at part load at lower temperatures with many operating hours. The choice of air cooled gascooler for supermarket systems could be supported by an analysis of the weighting of the operating conditions against the number of operating hours. One possibility is to use the same approach as for SEER (Seasonal Energy Efficiency Ratio) calculations employed in the European Eco Design requirements with a standardised temperature and load profile.

For the heat recovery and heat pump operation it is also clear that the default control setting needs to be changed at commissioning and it would greatly ease the work if the controller had a build in automated feature for this.

Further work is needed both to improve control systems but also to analyse in field studies how the actual performances of systems are and what the potential improvements could be.

However, based on this study there are indications that depending on the application improvement of the efficiency 5% to 20% could be within reach as well as the reduced cost of installation by improved selection of the gascooler.

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